DYNAMIC PRESSURE MEASUREMENTS ON A MODEL TURBINE RUNNER AND THEIR USE IN PREVENTING RUNNER FATIGUE FAILURE

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ABSTRACT

Hydraulic turbine runners, which operate under a wide range of heads and outputs, are subjected to considerable dynamic forces at off design conditions. Therefore a detailed knowledge of mean and fluctuating pressures is required for a safe design of the runner. A special technique has been developed at EPFL-IMHEF Laboratory for Hydraulic Machines in Lausanne, Switzerland, to measure the pressures on the blade of a model runner with miniature piezo-resistive pressure transducers. It has been successfully used in the Voith Hydro Laboratory during the model test of a large Francis turbine. Typical results are presented and discussed, together with their transposition to prototype conditions and application in the dynamic stress and fatigue analysis of the runner.

INTRODUCTION

Large Francis turbines are often called upon to operate under unfavourable conditions. These include operation over a wide head range and at part load and overload. Whilst the operation of a well designed Francis turbine is typically smooth in the whirl free zone, at progressively off design conditions pressure pulsations throughout the hydraulic channels of spiral casing, stay vane wicket gate cascade, runner and draft tube become significant. Accordingly the runners of turbines which are designed to operate under a wide range of head and output are subjected to considerable dynamic forces which can lead to fatigue cracking, especially in case of large runners, which inevitably, are relatively flexible. Also, particularly for large turbines, the worth of efficiency is high and fine tuning of the hydraulic profile to obtain optimal pressure distributions is a necessary part of the design and development process. Computerised Fluid Dynamics CFD aid in the process of accurately determining the runner pressure distribution, but experimental verification is a bonus. However currently CFD cannot determine fluctuating pressures which have, in the past, of necessity been inferred from measurements of draft tube and spiral case pressure fluctuations or measured on the model using strain gauged blades. Strain gauged blades will give some measure of pressures but suffer from the inevitable interaction with the blade structural characteristics, which could lead to misleading results.

In this paper the authors describe a state of the art technique using miniature piezo-resistive pressure transducers embedded in the model runner blades, which has been successfully used to measure the mean and fluctuating pressures on the blades of Francis model runners. Typical results obtained in the tests on a model for a ns = 273 rpm kW m, 295 MW Francis turbine are

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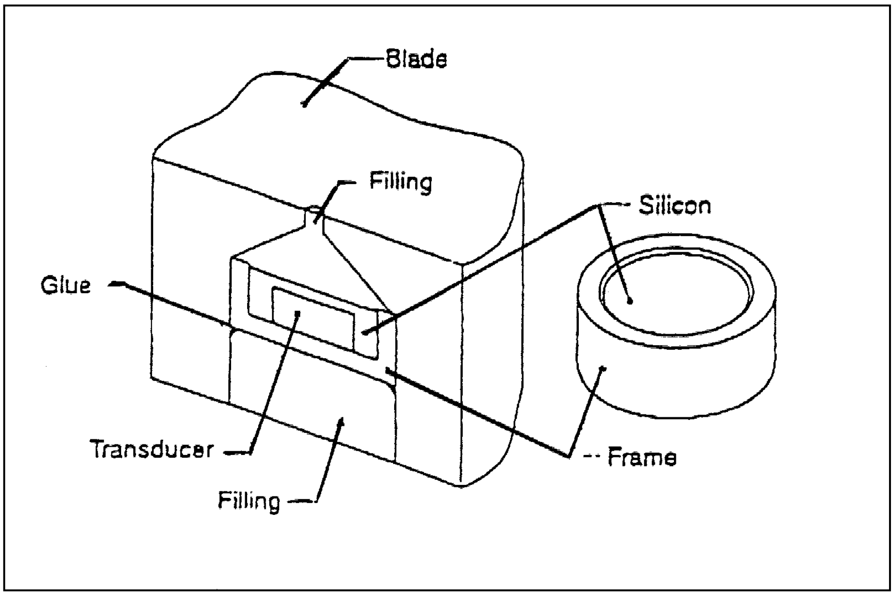
presented and discussed, together with their transposition to prototype conditions and application in the dynamic stress and fatigue analysis of the runner.

It is not proposed that this complex and expensive technique should be a standard procedure for model tests. However it has considerable merit to give a better understanding of pressure fluctuations in the runner and to provide basic data for progressive fine tuning and validations of numerical simulations.

INSTRUMENTATION AND TEST PROCEDURE

A special runner to be used only for the runner pressure measurements was constructed and the test rig and model, which had been used for performance testing, modified to accommodate it. The Keller miniature piezo-resistive pressure transducers were mounted inside the model turbine blade as shown in Figure 1.



A small hole connected the transducer with the fluid passage and the space filled with a plastic compound which had almost the same acoustic impedance as water. The space behind the transducer was filled with epoxy and the complete surface developed to the design blade profile.

Figure 1: Cross section of Keller transducer mounted in the runner blade

Photographs showing the mounting of the transducers and the finished blade with the transducers in place are given in Figure 2. For a typical 300 mm diameter model runner, only 6 transducers could be mounted on one individual blade. Hence, for a good distribution of measuring points on suction and pressure side of the blade four blades were instrumented and the outputs of the transducers electronically correlated in time with respect to the rotation of the model runner. The principal test instrumentation therefore comprised 6 transducers on the pressure side of two blades and six on the suction side of another two blades. Additionally, four transducers were mounted in the runner band and for cross correlation, two strain gauges were attached to the blades at the blade to runner crown junction. Four other blades were redundantly instrumented, to be used in the event of a major loss of transducers in the test blades.

Instrumented blades were bolted to the runner crown and band and the transducer wires were fed out to be collected on the model runner crown as shown in the photograph of Figure 3.

Figure 2: Photographs showing transducers mounted in the runner blade and the finished blade

The wires were fed to a signal conditioning system which was mounted on an annular chamber rotating with the runner. Eight acquisition modules mounted on a special shaft bearing system could handle 32 channels and had sufficiently memory to store 32768 samples per channel at a maximum frequency of 20 kHz and a 12 bits resolution. This represented more than 1 second sampling time. The acquisition modules were controlled through slip rings via a high level transfer protocol (ARCnet) by the laboratory central computer. After acquisition the data were downloaded to the laboratory central computer for further processing.

Figure 3: Photograph of the runner crown with data collecting modules

The pressure range of the transducers was 0 to 3 bars, frequency range was approximately 0 to 15 kHz and their sensitivity was 100 mV/bar for a current supply of 4 mA. Each transducer was 4.5 mm in diameter and 2.5mm thick. Angular position was monitored by two optical sensors mounted at 90^o on the model turbine shaft. Static calibration of the transducers between 0.3 and 2.5 bar against a high precision pressure transducer was conducted in the EPFL Laboratory by mounting the runner in a rotating water pressure chamber [1]. Verification of the static calibration was made in the Voith Hydro test rig by pressurising the rig in stages. Dynamic calibration of the transducers was made in the EPFL-Laboratory using a cavitation generator and a Kistler high precision transducer.

Obviously the scope of tests had to be kept to a reasonable number of operating points, commensurate with the availability of the test rig and that of the associated operating staff together with due allowance for the considerable time needed for processing the test results. On the other hand, the number of test points had to be representative of the predicted operation of the unit, such that a weighted loading for each operating point could be allotted for inclusion in the computation of the load universe. The starting point for the choice of test points therefore was the specified description of operation of the turbine which was an integral part of the contractual weighted efficiency and the head frequency curve. Test points indicated by these sources were then further defined using the measurements of draft tube pressure pulsations, to determine the worst cases within those general limits, all mutually agreed between the turbine manufacturer and Engineer before the start of the tests. For the turbine under consideration the head frequency curve indicated that the majority of operation was between rated and maximum head. The contractual weighting for efficiency was as follows:-

Head	Output	Loading
Rated	Rated	30%
Rated	Best gate	60%
Rated	70% rated	10%

Accordingly these three points were natural choices to test, however by mutual agreement the third point was modified to the point of Maximum Head, Maximum Draft Tube Pressure Fluctuations. To accommodate low head operation, chosen for test were Minimum Head, Maximum Output and Minimum Head, Maximum Draft Tube Pressure Fluctuations. The contract specified survival for 30 minutes at maximum runaway speed which became the final test point for the load universe. Additional tests were conducted at one of the chosen part load points to see the effect, if any, of different suction heads (σ_{runner} and $\sigma_{draft\ tube}$) and air content. All other than the runaway speed test were conducted at cavitation free conditions.

All normal turbine test rig instrumentation was operative during the runner pressure tests other than the shaft torque measuring system which could not be used because of conflict with the runner pressure acquisition system. It was therefore mutually agreed that the operating point in respect of output would be determined via the wicket gate opening, previously established during the performance tests. Air content was accessed with a dissolved oxygen probe. Tests essentially consisted of establishing the operating turbine point and starting the on board acquisition system. After each test the acquired data were downloaded through the slip rings and then processed. Acquisition time was very short, however processing of results took up to 3 hours per test point. After each test the runner pressure transducers were corrected for drift.

Although the signals from some transducers were lost during the test series enough remained functioning to establish a representative fluctuating pressure field. As expected most of the damage occurred during the runaway speed test as a result of cavitation. For this reason the test at runaway speed was conducted at the end of the test series. The redundant transducers were not used.

RESULTS AND INTERPRETATION OF MEASUREMENT

A typical measured mean pressure distribution across the suction and pressure side of the runner blades is shown in Figure 4 and the comparison with the corresponding Computerised Fluid Dynamic (CFD) calculation using a Navier Stokes k-ε model in Figure 5.

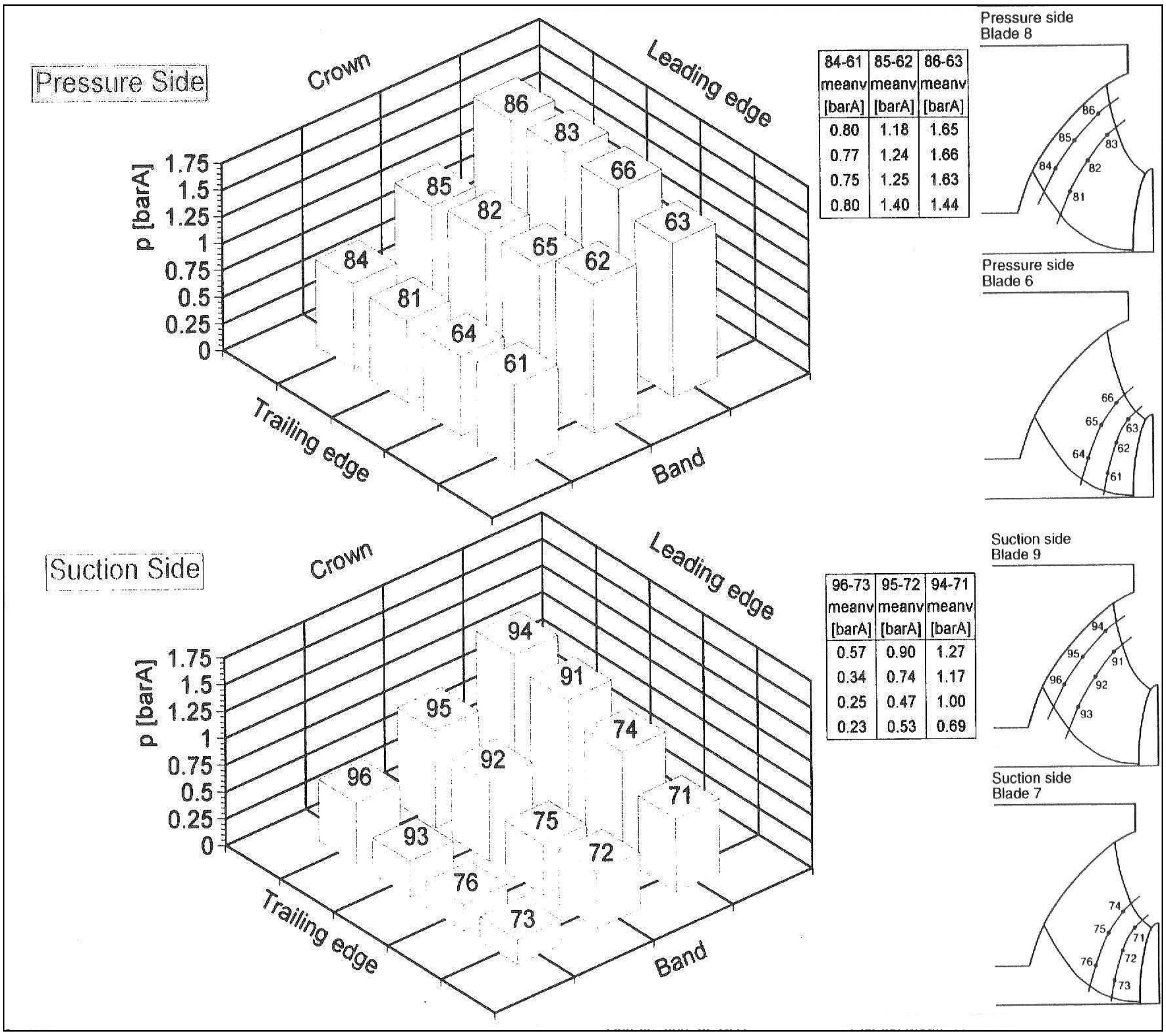


Figure 4: Mean pressure distribution for H_{rated} P_{rated}

Agreement between the measured and calculated model pressures was good, giving confidence in the accuracy and subsequent validity of the measuring method and results.

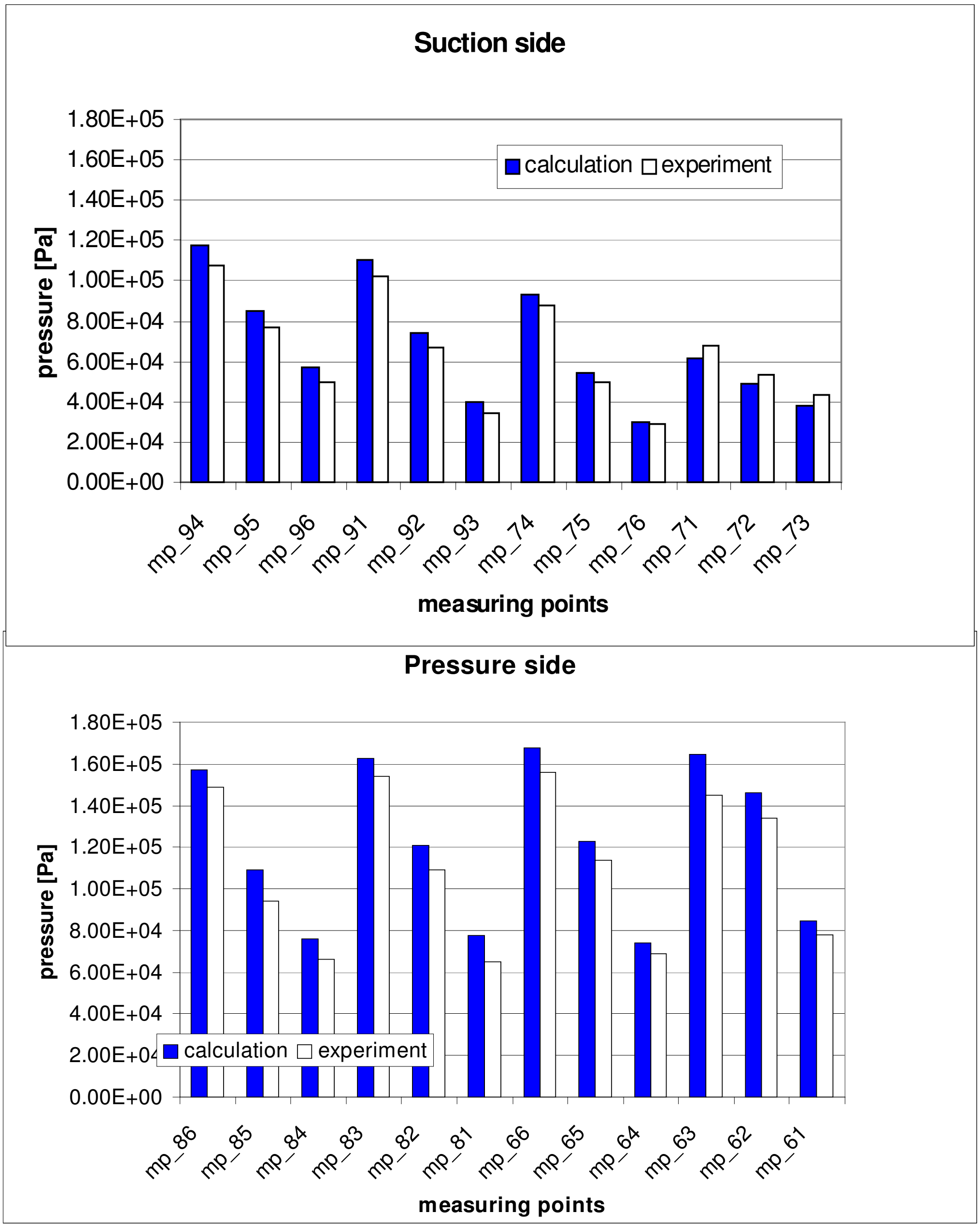


Figure 5: Comparison of measured mean pressures with those calculated for H_{max} and best gate

The waterfall diagram of measured pressure fluctuation in the runner for the condition H_{min} and maximum draft tube pressure fluctuations is given in Figure 6 based on a model test Head of 14.148m. This operating condition gave the maximum runner fluctuating pressures and was chosen to investigate the effect of different σ's and model speed on the measured pressures. The effect of change of model speed was negligible when the results were adjusted for the change in model head, necessary for equal model unit speed for the two conditions. As expected fluctuating pressures were slightly higher with σ_{runner}, which was approximately 13% less than $\sigma_{draft\ tube}$. However it was agreed that, as the runner pressure fluctuations for part load operation reflect draft tube conditions, the runner pressure fluctuations corresponding to $\sigma_{draft\ tube}$ would be used in the runner stress computations. With similar reasoning the results with

σ_{runner} were used for the near design conditions, where draft tube flow conditions are optimal. The degree of air content had no effect on the measured pressures.

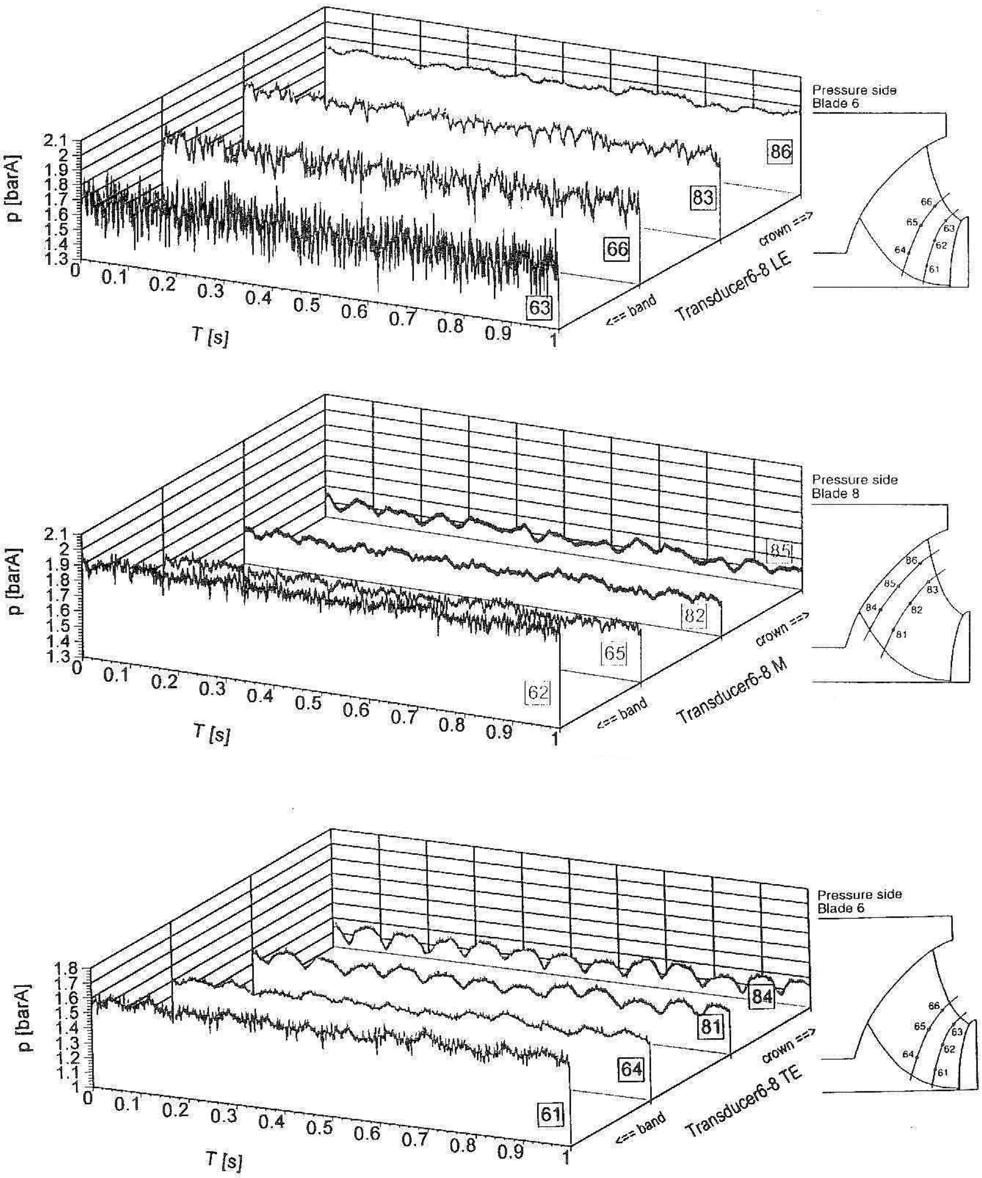


Figure 6: Waterfall diagram of measured pressure fluctuation in the model runner for the condition H_{min} and maximum draft tube pressure fluctuations

The frequency spectra for suction and pressure side fluctuating pressures are given in Figure 7 for the condition H_{min} and the gate opening for maximum draft tube pressure pulsations. Model rotating speed was 17 Hz and peaks in the spectrum are apparent in most plots at:-

- Synchronous draft tube vortex frequency - about 1/3 rotational frequency or 5.6 Hz
- Relative draft tube vortex frequency – about 2/3 rotation frequency or 11.4 Hz
- Rotation frequency – 17 Hz
- The higher harmonics of relative draft tube vortex frequency
- The higher harmonics of rotational frequency
- Wicket gate passing frequency (24 rotational frequency)

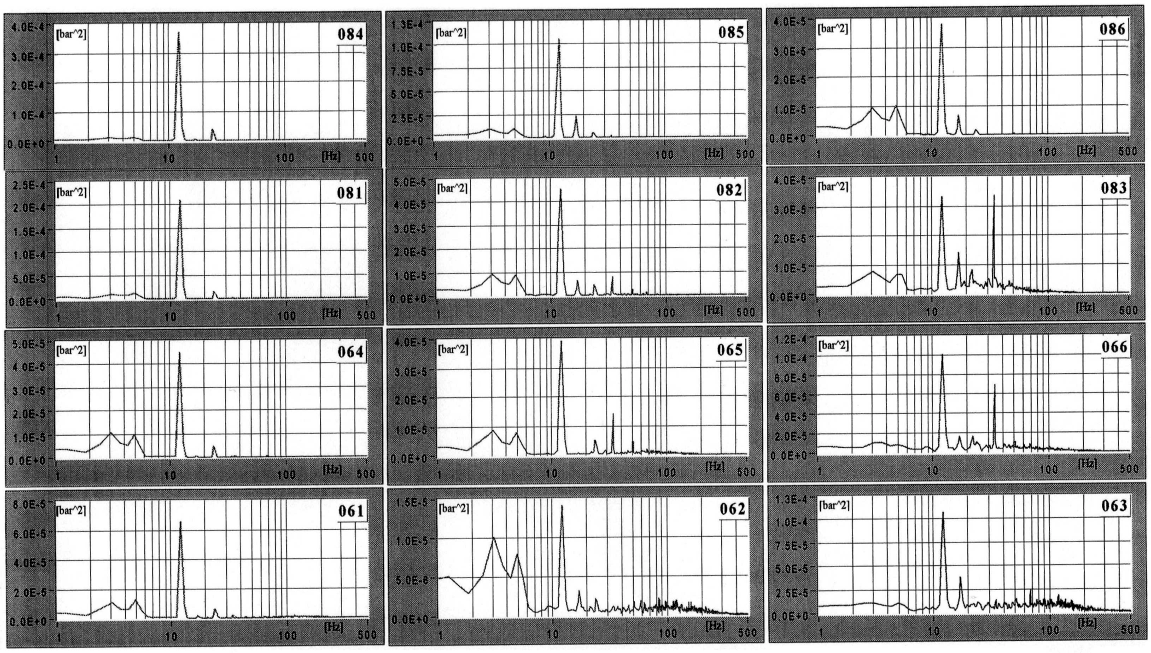


Figure 7: Frequency spectrum plots for the condition H_{min} and maximum draft tube pressure fluctuations (dominant peak at relative draft tube vortex frequency)

At part load the dominant runner pressure pulsation is at the relative draft tube vortex frequency and has a maximum amplitude of about ±2.8% H_{min} on the blade suction side at the discharge and ±1.4% H_{min} on the pressure side. At runner inlet the corresponding values are ±0.84% H_{min} on the suction side and ±0.71% H_{min} on the pressure side. Maximum fluctuating pressures at the discharge occur towards the band on the suction side and towards the crown on the pressure side. At the runner inlet, maxima occur on both the suction and pressure side towards the band. As a comparison, the draft tube pressure amplitude measured on the tailwater side of the discharge cone for this operating condition was ±4.4% H_{min}. From Figure 8 it is seen that part load operation at H_{max} (model test Head of 22.385m) gave runner pulsation amplitude of ±1.36% H_{max} on the blade suction side at the discharge and ±0.6% H_{max} on the pressure side. Maximum amplitude of runner pulsations at the inlet for operation at H_{max} were ±1.84% H_{max} on the suction side and ±0.34% H_{max} on the pressure side. The corresponding amplitude of draft tube pressure pulsations measured at the cone on the tailwater side was ±3.55% H_{max}.

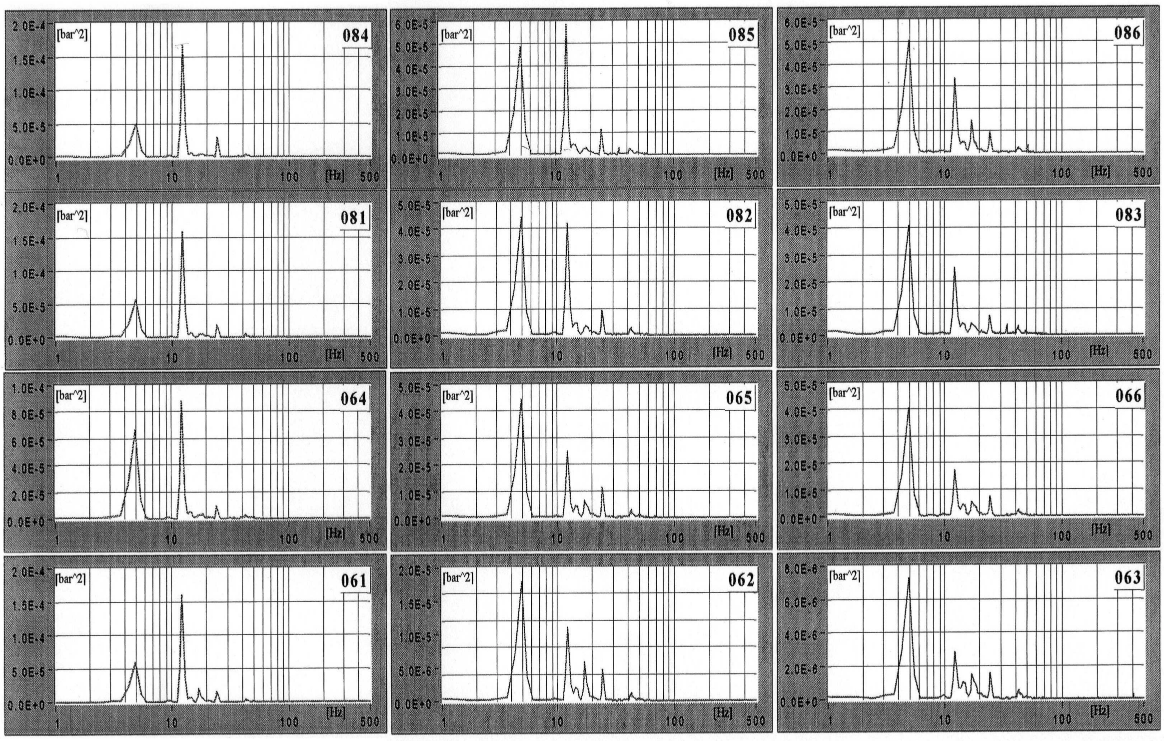


Figure 8: Frequency spectrum plots for the condition H_{max} and maximum draft tube pressure fluctuations (dominant peaks at synchronous and relative draft tube vortex frequency)

Similar frequency spectrum curves are given for H_{rated}, best gate (model test Head of 21.208m) in Figure 9 from which it is evident that the dominant frequency for this operating condition is that corresponding to the rotational speed and its harmonics. Maximum runner pressure fluctuations are on the suction side on the discharge and are approximately 30% of those measured in part load operation at H_{min}, when corrected for head.

The runaway speed test gave essentially random distributions of fluctuating pressure in the frequency domain with a marked peak however at 40Hz (which is about 2.3 times of rotating speed), especially on the pressure side. It is thought that this peak results from a central vortex in the draft tube at about 24 Hz (1.35 times rotational frequency) rotating contra to the runner rotation. This central vortex would be similar to that seen at gross overload and differs from the part load vortex in both rotational frequency and direction of rotation relative to the runner. There was confirmatory evidence of a central vortex rotating at 24 Hz in the measured radial thrust and spiral pressure fluctuations at runaway.

RUNNER FATIGUE LIFE STUDY

Transposition of the results to the prototype was straight forward. All frequencies were scaled in the ratio of model to prototype rotational speed. For all, other than the relative draft tube vortex frequency at part load condition, fluctuating pressures were used in the analysis in the form of the ratio of fluctuating pressure and mean pressure.

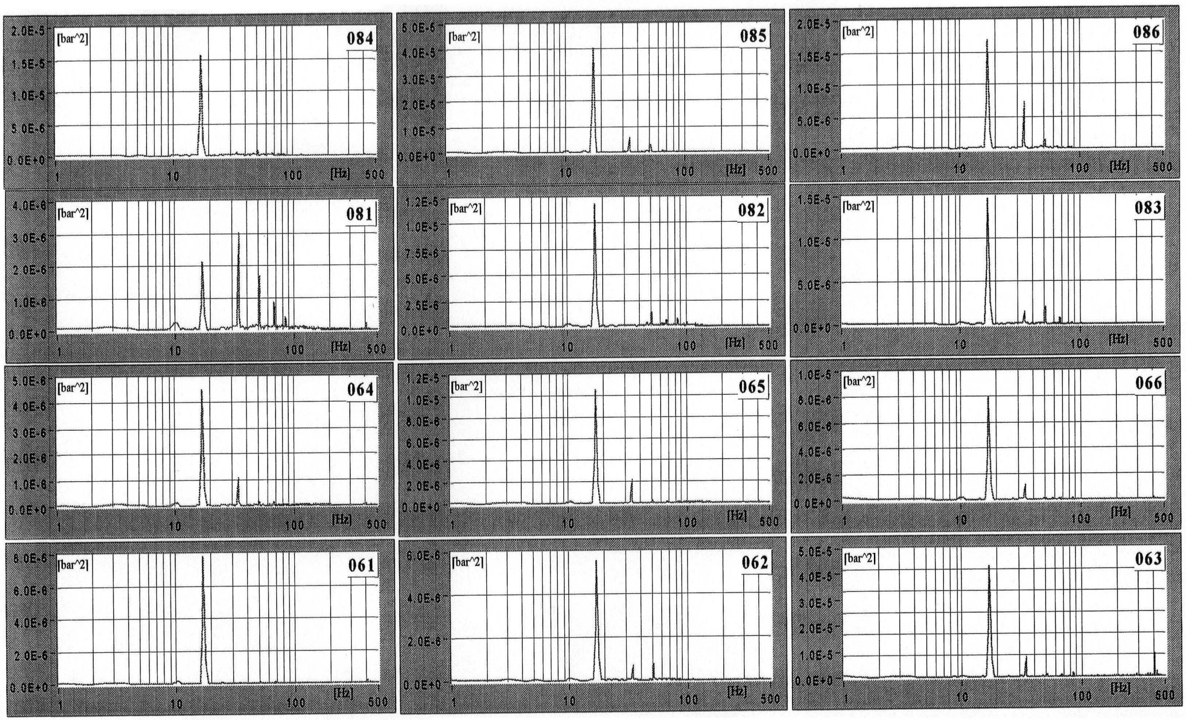


Figure 9: Frequency spectrum plots for the condition H_{rated} best gate (dominant peaks at rotation speed and harmonics)

This was so that the fluctuating pressures could be superimposed upon the finite element stress analysis of the runner already conducted for that particular operating condition using the mean pressures. The static pressures were obtained from the CFD analysis for all conditions other than runaway where the CFD calculation did not converge. Hence for runaway an interpolation was made of the measured mean pressures. The interpolation was necessary because of the non functioning pressure transducers. For the particular case of part load operation a pressure distribution for the relative draft tube vortex frequency was derived from the measured fluctuating pressures and applied to the Finite Element model as a special load case. All pressures were adjusted for the difference in model and prototype suction head and then scaled to prototype values in the ratio of the model to prototype net heads.

A fundamental problem for the fatigue analysis is how to derive dynamic stresses from the measured fluctuating pressures. This was a major decision to be made between the turbine manufacturer and the Engineer. The measured fluctuating pressures were complex in terms of frequency and amplitude and could be used in a variety of ways to estimate the dynamic stresses. However it was recognised by both parties that the end result would of necessity only be a coarse estimate of reality, representing a worst case scenario. Accordingly a very conservative approach was adopted which was acknowledged to lead to potentially unrealistically high stresses. The philosophy being that, if the runner had acceptable life for these assumptions then no further sophistication was required, if not, then a more precise method should be used.

Accordingly the maximum ratio of fluctuating pressure to mean pressure was taken at each frequency of note (draft tube vortex, rotational speed, harmonics etc). Dynamic stress was then obtained by multiplying the maximum static FEM stress by this ratio. Stresses thus obtained were further adjusted by a dynamic gain factor calculated from a modal analysis of the runner structure. Fatigue life of the runner under these stresses was calculated according to the Palmgren-Miner theory. Using the Haigh Diagram for runner material with water influence the number of cycles to failure was obtained. A life of 50 years was assumed and for each frequency of note the number of operating cycles was obtained, using time weightings corresponding to the specified mode of operation of the generating unit. The ratio of cycles to failure and actual estimated cycles for the particular frequency of note gave the safety margin in terms of damage accumulation. The final result indicated a virtual infinite life for the runner.

The possibility of growth of weld defects during operation was investigated with fracture mechanics in conjunction with typical allowable defect size. The dynamic stresses used were those previously established for the Palmgren-Miner analysis. Results of this analysis showed that any defects that may exist in the high stressed region would not propagate to any significant size in respect of operation of the runner, over a period considerably longer than the power station design life.

CONCLUSIONS

The tests described above have established that measurement of runner blade mean and fluctuating pressures can now be conducted in a commercial laboratory as part of a contractual model turbine performance test. Additionally, with proper agreement between all parties, the results obtained can be used in a meaningful analysis of runner fatigue life. There is an intention at Voith Hydro to conduct similar tests on models of reversible pump turbines and other Francis turbine models which have more difficult operating ranges than the subject of this paper. Results from these tests will be evaluated systematically to build a compendium of data for general use on other projects. However it again must be emphasised that these are special tests which are only warranted for large turbines which operate under large head ranges and difficult part load conditions. They are definitely not required for established design parameters for standard turbines which operate with normal operating limits.

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